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Miguel E. Jovane
Purdue University

James E. Braun
Purdue University

Eckhard A. Groll
Purdue University

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Miguel E. Jovane*, James E. Braun, Eckhard A. Groll

Ray W. Herrick Laboratories
Purdue University
West Lafayette, Indiana 47907, USA

*corresponding author: jovane@purdue.edu

ABSTRACT

Design of evaporator coils or other heat exchangers always involves a tradeoff between heat transfer and pressure drop. One idea that is considered in this paper for changing this tradeoff involves extracting refrigerant vapor from the two-phase, evaporating flow stream. It was hoped that an extraction strategy could be identified that would lead to reduced heat transfer requirements for the same cooling capacity and pressure drops as for a conventional coil. In order to evaluate this concept, an analytical model was developed that allows continuous extraction of the vapor from the two-phase flow stream. The refrigerant, the mass flow rate, the geometry of the coils, the number of circuits, and the inlet quality of the refrigerant are the input parameters of the model. The model is described in this paper and example results are presented for different extraction strategies using refrigerant R410A as the working fluid. For all the cases studied, the extraction of vapor did not provide any positive benefits in terms of reducing the required heat transfer area necessary to realize a given capacity and pressure drop.

1. INTRODUCTION

One approach for extracting vapor from an evaporator during the evaporation process would be to utilize return bends as phase separators as depicted in Figure 1. A certain fraction of the saturated vapor would be separated from the two-phase mixture at multiple return bends along the flow paths and mixed with refrigerant exiting the evaporator. Prior to this study, it was postulated that vapor extraction could lead to smaller heat exchanger requirements for a given capacity and pressure drop. This paper evaluates this postulate for some different strategies of extracting vapor from the two-phase flow stream of an evaporator using a detailed model. The model solves coupled momentum and energy equations. The coupled system of equations that describe an evaporator coil is solved numerically for two cases: one where the pressure gradient is specified using the pressure drop of an evaporator with no extraction as reference; and another where the rate of extraction (i.e. mass flow rate profile) is specified. In both cases, the cooling capacity, pressure drop, tube diameter for the coil were held constant and length the tubing required was determined. In order to achieve the same pressure drop, the number of circuits used in the extraction cases was smaller than the number of circuits of the reference (no extraction) case. The constraints of constant capacity and pressure drop, imply that the total inlet mass flow rates and the exit states for all extraction cases were the same as those for the reference case. The results show a clear adverse effect of the vapor extraction process on the two-phase heat transfer coefficient. For all cases refrigerant R410a was used as the working fluid.

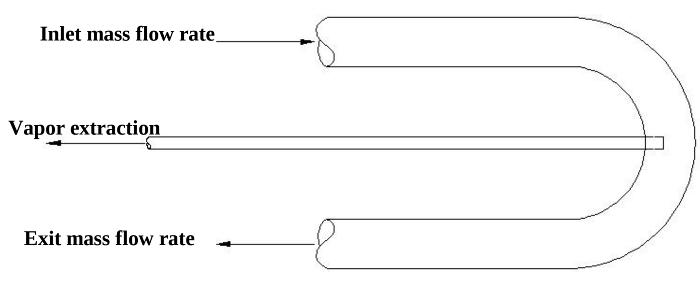

Figure 1. Vapor extraction process

2. MODEL OF PHASE SEPARATION

An evaporator model was developed to determine the required tube length necessary to achieve a specified refrigerant exit state for given inlet conditions and flow rate with different vapor extraction strategies. It is assumed that the vapor is continuously extracted from the stream and that the process occurs at steady state. In practice, vapor extraction could be implemented at the return bends for each individual tube pass within the evaporator. The evaporator tubes are assumed to be oriented horizontally so that gravitational effects are negligible. Other assumptions include one-dimensional flow, constant cross sectional area, and smooth tubes. Conservation of mass, momentum, and energy are employed. Only a single tube is modeled and it is assumed that parallel flow paths have identical boundary conditions.

2.1. Conservation of mass

A differential control volume for the refrigerant flow stream is depicted in Figure 2 with continuous extraction (or addition) of vapor. Applying conservation of mass to the differential control volume of Figure 2, the mass flow rate of vapor added to the streams is given by,

$$\frac{\delta \dot{m}}{\delta z} = \frac{d\dot{m}}{dz} \tag{1}$$

2.2. Conservation of linear momentum

The steady-state conservation of linear momentum for a one-dimensional two-phase mixture with negligible body forces can be written as,

$$\frac{d}{dz}\left(\dot{m}_g u_g + \dot{m}_f u_f\right) - u_g \frac{d\dot{m}}{dz} = -\frac{d}{dz}\left[P\left(A_g + A_f\right)\right] - \left(\tau_f p_f + \tau_g p_g\right) \tag{2}$$

where P is the pressure, τ is the shear stress, p is the tube internal perimeter, u is velocity, and the subscripts f and g refer to saturated liquid and vapor conditions. In developing Equation (2) it has been assumed that the mass added or extracted is in contact only with the vapor inside the tube as depicted in Figure 3.

2.3. Conservation of energy

Neglecting potential energy changes, the conservation of energy equation is given by,

$$\frac{d}{dz}\left[\dot{m}\left(x h_g^0 + (1-x) h_f^0\right)\right] - h_g \frac{d\dot{m}}{dz} = q'' p \tag{3}$$

where h^o is the stagnation enthalpy $h+u^2/2$, x is the flow quality, and q" is the heat flux.

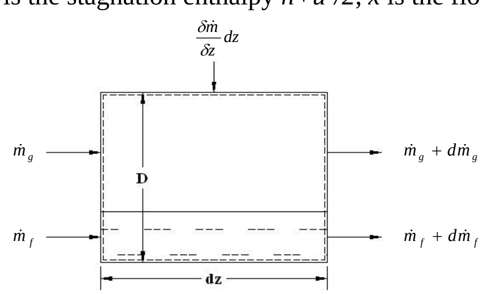

Figure 2: Conservation of mass

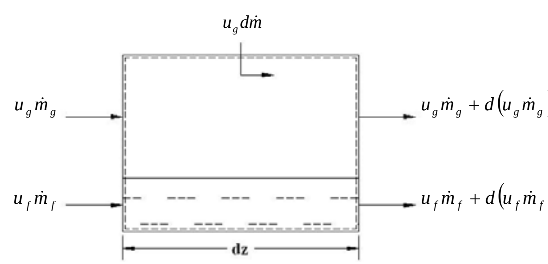

Figure 3. Rate of change of momentum

2.4. Solution

Equations (2) and (3) can be coupled and expressed as a function of the rate of change of the pressure, quality, and mass flow rate through,

$$\begin{bmatrix} M_P & M_x & M_{\dot{m}} \\ E_P & E_x & E_{\dot{m}} \end{bmatrix} \begin{bmatrix} \frac{dP}{dz} & \frac{dx}{dz} & \frac{d\dot{m}}{dz} \end{bmatrix}^T = \begin{bmatrix} \left(\frac{dP}{dz}\right)_f \\ q'' p \end{bmatrix} \tag{4}$$

where, $\left(\frac{dP}{dz}\right)_f$ is the frictional pressure drop, and the coefficient of the 2 by 3 matrix are,

$$M_P = 1 + \frac{\dot{m}^2}{A^2}\left[\frac{x^2}{\alpha}\frac{dv_g}{dP} + \frac{(1-x)^2}{1-\alpha}\frac{dv_f}{dP} + \left(-\frac{x^2 v_g}{\alpha^2} + \frac{(1-x)^2 v_f}{(1-\alpha)^2}\right)\frac{\partial\alpha}{\partial P}\right] \tag{5}$$

$$M_x = \frac{\dot{m}^2}{A^2}\left[2\left(\frac{x v_g}{\alpha} - \frac{(1-x)v_f}{1-\alpha}\right) + \left(-\frac{x^2 v_g}{\alpha^2} + \frac{(1-x)^2 v_f}{(1-\alpha)^2}\right)\frac{\partial\alpha}{\partial x}\right] \tag{6}$$

$$M_{\dot{m}} = \frac{\dot{m}}{A^2}\left[\frac{x(2x-1)v_g}{\alpha} + \frac{2(1-x)^2 v_f}{1-\alpha} - \dot{m}\left(\frac{x^2 v_g}{\alpha^2} - \frac{(1-x)^2 v_f}{(1-\alpha)^2}\right)\frac{\partial\alpha}{\partial\dot{m}}\right] \tag{7}$$

$$E_{\dot{m}} = (x-1)h_{fg} + \frac{3}{2}\frac{\dot{m}^2}{A^2}\left[\frac{x^3 v_g^2}{\alpha^2} + \frac{(1-x)^3 v_f^2}{(1-\alpha)^2}\right] - \frac{\dot{m}^3}{A^2}\left[\frac{x^3 v_g^2}{\alpha^3} - \frac{(1-x)^3 v_f^2}{(1-\alpha)^3}\right]\frac{\partial\alpha}{\partial\dot{m}} \tag{8}$$

$$E_x = \dot{m}h_{fg} + \frac{\dot{m}^3}{A^2}\left[\frac{3}{2}\left(\frac{x^2 v_g^2}{\alpha^2} - \frac{(1-x)^2 v_f^2}{(1-\alpha)^2}\right) + \left(-\frac{x^3 v_g^2}{\alpha^3} + \frac{(1-x)^3 v_f^2}{(1-\alpha)^3}\right)\frac{\partial\alpha}{\partial x}\right] \tag{9}$$

$$E_P = \dot{m}\left(x\frac{dh_g}{dP} + (1-x)\frac{dh_f}{dP}\right) + \frac{\dot{m}^3}{A^2}\left[\frac{x^3 v_g}{\alpha^2}\frac{dv_g}{dP} + \frac{(1-x)^3 v_f}{(1-\alpha)^2}\frac{dv_f}{dP} - \left(\frac{x^3 v_g^2}{\alpha^3} - \frac{(1-x)^3 v_f^2}{(1-\alpha)^3}\right)\frac{\partial\alpha}{dP}\right] \tag{10}$$

In Equations (5) through (10), α represents the void fraction which is a function of the pressure, quality, and mass flow rate. For specified geometrical parameters, inlet conditions, and mechanistic equations for the frictional pressure drop, heat flux, and void fraction, the number of unknowns is greater than the number of equations. It is also necessary to specify a strategy for extracting vapor from the flow stream. Some possible strategies include constant mass flow rate (no extraction), extraction to achieve constant refrigerant quality ($dx/dz=0$), extraction rates that achieve specified variations in mass flow rate, pressure, or quality, etc.

The heat transfer rate per unit of length $q''p$ of equation (4) is predicted using,

$$q' = q''p = h_{tp}\,p\left(T_{wall} - T_{ref}\right) \tag{11}$$

where h_{tp} is the refrigerant two-phase heat transfer coefficient and T_{wall} is the inner tube wall temperature, and T_{ref} is the refrigerant temperature. The wall temperature is unknown and in order to estimate the heat flux, Equation (11) needs to be coupled with the air side of the evaporator. Neglecting the thermal resistance between the inner and outer wall of the tube, the heat transfer rate per unit of length for the air side point of view is expressed as,

$$q' = UA'_{air}\left(T_{air} - T_{wall}\right) \tag{12}$$

where UA'_{air} is the air side thermal conductance per unit of length and T_{air} is the air temperature that is assumed constant in this study. An air side thermal conductance per unit of length of 19.25 W/m-K was assumed. This conductance is in acceptable agreement with the predicted conductance of evaporator coils of similar capacity. The tube inside diameter is assumed equal to 9 mm. The refrigerant two-phase heat transfer coefficient is predicted using a regime dependent correlation. The regime was predicted using the map proposed by Wojtan *et al.* (2005) and the heat transfer coefficient was predicted using a relation as proposed by Kattan *et al.* (1998) introducing the modifications suggested by Wojtan *et al.* (2005). This relation is given by,

$$h_{tp} = \frac{\theta_{dry}h_v + \left(2\pi - \theta_{dry}\right)h_{wet}}{2\pi} \tag{13}$$

where the angle θ_{dry} depends on the regime. Both Kattan and Wojtan suggest using the modified Rouhani-Axelsson correlation proposed by Steiner (1993) for estimating the void fraction. This relation is,

$$\alpha = \frac{x}{\rho_g}\left[\left(1+0.12(1-x)\right)\left(\frac{x}{\rho_g}+\frac{1-x}{\rho_f}\right)+\frac{1.18(1-x)\left[g\sigma\left(\rho_f-\rho_g\right)\right]^{0.25}}{G\rho_f^{0.5}}\right]^{-1} \qquad (14)$$

In this study, this correlation was used because of its accuracy as reported by Wojtan et al. (2005) and because it accounts for the effect of the mass flux on the void fraction. The pressure drop due to friction was calculated using the correlation proposed by Müller-Steinhagen and Heck (1986).

3. RESULTS

Using the model presented in the previous section, the system of Equations (4) was solved for cases involving both no mass extraction (constant mass flow rate) and different extraction strategies. Refrigerant R410A was used as the working fluid and the inlet conditions and mass flow listed in Table 1 were employed. The reference case was assumed to be for constant refrigerant mass flow rate with the flow split into 5 parallel circuits. The configuration, tube geometry and boundary conditions are consistent with a 3-ton unit that uses refrigerant R410a. For the reference case, the system of equations was solved using Euler's explicit method and the length of tubing was determined that yielded an exit state of refrigerant that was a saturated vapor. This solution established a reference exit state, a reference pressure distribution, and a reference tube length for use and comparison with the different extraction strategies. Some results for the reference case are tabulated in the first row of Tables 2 and 3. These tables also give results for the two extraction cases described in the following two subsections. The extraction cases required fewer parallel circuits than the reference case in order to achieve the same exit pressure, exit quality which and cooling capacity which was approximately 10 kW.

4.1. Specified pressure variation

The pressure distribution obtained for the reference case with no mass extraction was curve-fit as a function of a relative position defined as the absolute position, *z*, along the flow path divided by the tube length for each individual circuit, *l*. The curve fit was then used to specify the pressure gradients for other cases where vapor was removed (or added) from the flow stream. Cases with three and four parallel circuits were considered. For each case studied, the tube length necessary to achieve a saturated vapor state with the reference exit pressure was determined. Results are shown in Table 2. In both cases, the total length of tubing required to achieve the reference case exit condition was greater than the reference tube length. Figures 3, 4, and 5 show refrigerant pressure, two-phase heat transfer coefficient, mass flow rate, and quality as a function of relative length along the tube for the different cases. The pressure variations for the extraction cases (3 and 4 circuit) are constrained to be within 0.1% of the reference pressure distribution according to the curve fit. At the inlet to the tube, the heat transfer coefficients for both extraction cases are greater than that for the reference case due to the greater mass flow rate per circuit. However, as soon as vapor is removed from the stream, the heat transfer coefficient drops and quickly falls below the reference case where it stays for most of the flow path. It is observed in Figure 6 that the rate of quality increase for the cases where vapor is extracted is slower than that of the reference case. This can be expected since vapor is extracted from the stream for most of the path. This also affects the heat transfer coefficient since for a two-phase mixture the heat transfer coefficient is proportional to the flow quality until dryout conditions are achieved. The decrease in the heat transfer coefficient is the reason why the total tube length required for the cases where vapor is removed is greater that the total tube length of the reference case.

Table 1. Inlet conditions

Inlet pressure [kPa]	930.862
Inlet quality	0.2059
Total mass flow rate [kg/h]	42.00
Air temperature [^{o}C]	18.78

Table 2. Results for the similar relative pressure pattern

# of circuits	Tube length per circuit [m]	Total tube length [m]	Exit pressure [kPa]	Exit quality	Inlet flow rate per circuit [kg/h]	Exit flow rate per circuit [kg/h]
5 (reference)	9.106	45.530	921.586	0.999	42.00	
4	11.575	46.300	921.491	0.999	52.50	39.07
3	15.859	47.577	921.491	0.999	70.00	31.64

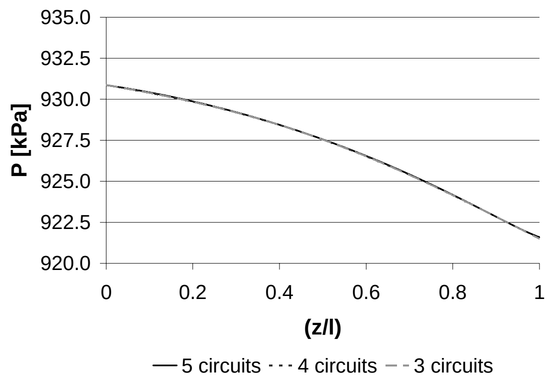

Figure 4. Pressure profile

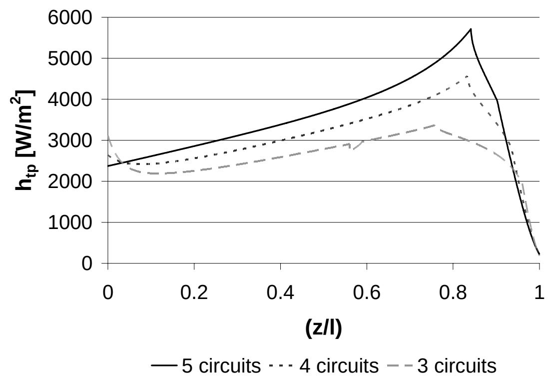

Figure 5. Two-phase heat transfer coefficient

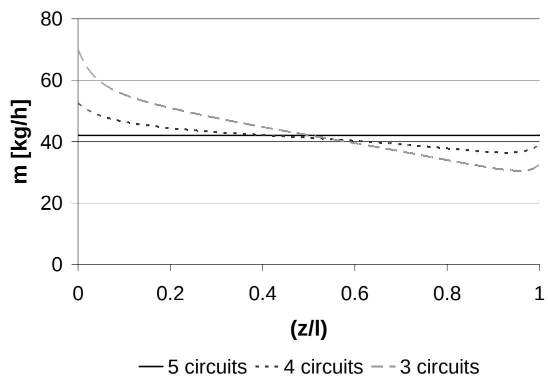

Figure 6. Mass flow rate per circuit

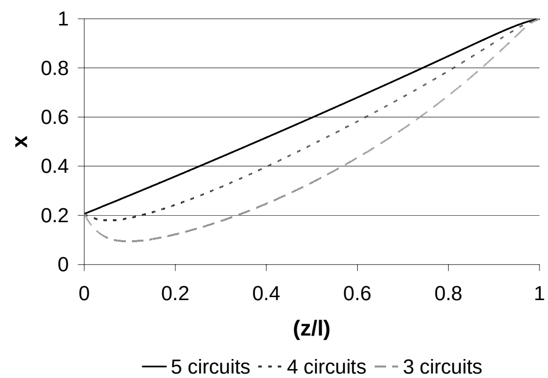

Figure 7. Flow quality

Several other possible pressure drop profiles can be prescribed. For example, a linear pressure variation along the refrigerant flow path for four and three circuits was investigated but led to a requirement for vapor addition at the beginning of the evaporation process. This would be difficult to implement physically since the only vapor available for addition would be vapor extracted downstream of the coil, and this vapor would have a pressure lower than the pressure required at the beginning of the evaporation process. From the linear pressure distribution study it was also observed that total tube length required for the cases where vapor was allowed to leave or enter the coil was greater than the length of the reference case. The difficulties of adding vapor to the coil rather than extracting it led to the idea of specifying a mass extraction strategy along the coil instead of a pressure distribution strategy.

4.2. Specified mass flow variation

Several different strategies were investigated and examples are presented here where mass was extracted following a quadratic profile. For these examples, vapor was extracted until a specific coil exit mass flow rate was reached. Then the mass flow rate was kept constant for the remainder of the flow within the evaporator. The coefficients of the quadratic mass flow rate function were determined using the coil inlet and outlet mass flow rates as boundary conditions. The third coefficient of the quadratic expression was determined by assuming that the mass flow rate reaches the exit flow in a smooth way (no gradient). The exact point along the tube at which the exit mass flow rate was reached was determined iteratively by matching the exit pressure to the exit pressure for the reference case (constant mass flow rate with 5 circuits). There are upper and lower bounds for the coil exit mass flow rate for this extraction strategy. The lower limit for coil exit mass flow rate occurs for the case where the quadratic function applies over the entire length of the coil while achieving the required exit state. For other cases, it was observed that as the exit mass flow rate is increased, it is required to extract the vapor earlier in the evaporation process in order to achieve the exit pressure. Since at the beginning of the process the available amount of vapor is small, the upper limit for coil exit mass flow rate is associated with achieving a minimum refrigerant quality of zero (i.e. extracting the entire available vapor early in the evaporation process) at the end of the extraction process. If a greater exit mass flow rate is to be achieved vapor would have to be added to the coil. Table 3 and Figures 8 to 11 show results for these two limiting cases with a four circuit configuration in comparison to the reference case. It is seen from Table 3 that the total length for the two cases presented is greater than the total length of the reference case. Figure 8 shows that the average two-phase heat transfer coefficient is smaller than that of the reference case. This is true even for the case where the extraction is stopped at a mass flow rate that is 97.8% (upper limit) of the reference mass flow rate. For this particular case, Figure 10 shows that in order to reach the reference exit pressure the extraction process must be stopped early in the process. Figure 11 shows that more vapor than what is produced is extracted from the stream resulting in a decrease in quality. Since the two-phase heat transfer coefficient is directly proportional to the flow quality until dryout conditions are reached, delaying the increase of the flow quality has a decreasing effect on the heat transfer coefficient. Although this effect is amplified for the case where the exit flow rate is 97.8% of the reference flow, the flow quality increased at a smaller rate for all cases where vapor was extracted in this study.

Table 3. Quadratic mass flow profile (4 circuits)

Exit flow rate [% of reference mass flow]	Tube length per circuit [m]	Total tube length [m]	Exit pressure [kPa]	Exit quality	Inlet flow rate per circuit [kg/h]	Exit flow rate per circuit [kg/h]
100 (reference)	9.106	45.530	921.586	0.999	42.00	
85.4	11.565	46.260	921.586	0.999	52.50	35.88
97.8	11.646	46.584	921.586	0.999		41.08

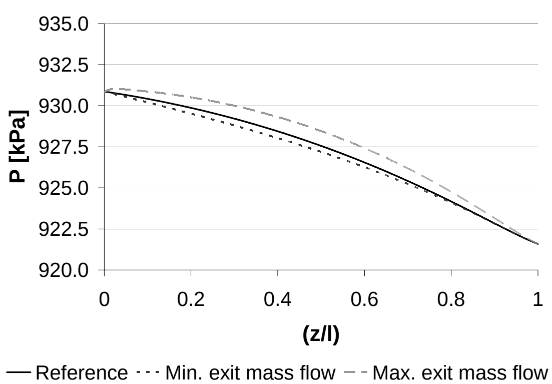

Figure 8. Pressure distribution (4 circuits)

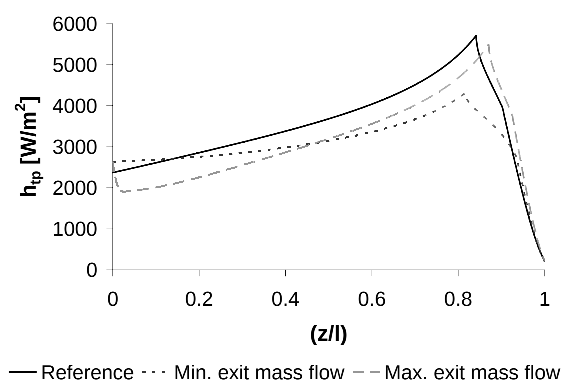

Figure 9. Two-phase heat transfer coefficient (4 circuits)

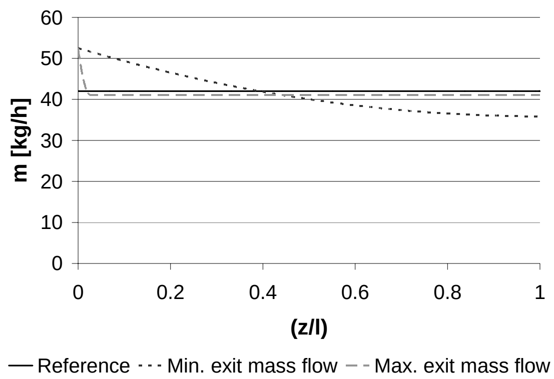

Figure 10. Mass flow rate per circuit (4 circuits)

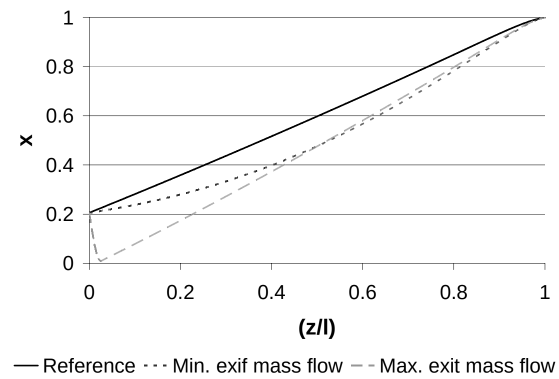

Figure 11. Flow quality (4 circuits)

Several other four circuit cases were studied using a quadratic mass flow rate extraction strategy between lower and upper limits on exit mass flow rate. Figure 12 shows the required relative tube length for all cases considered. The relative tube length was generated by dividing the total tube length for each case by the total tube length of the reference case. Figure 12 clearly shows that for all cases the total tube length required is greater than the reference tube length.

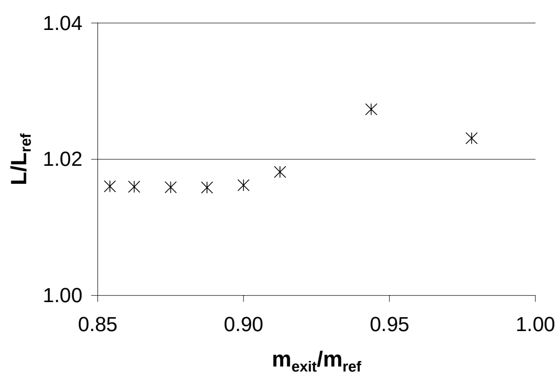

Figure 12. Total tube length for the quadratic mass extraction (4 circuits)

It is important to mention that a range of other extraction strategies were studied. In all cases considered, it was observed that the extraction process always had a negative effect on the required tube length.

4. CONCLUSIONS

From the model developed and the cases studied, it can be observed that extracting vapor from the two-phase flow stream of an evaporator coil is not a good strategy for reducing required heat transfer area. This is due to the combined effect of the reduction in the mass flow rate and the decrease in the rate of change of the flow quality. These two factors directly affect the two-phase heat transfer coefficient. A smaller mass flow rate leads to a smaller heat transfer coefficient, and a decrease in the rate of change in the flow quality also has a decreasing effect on the heat transfer coefficient because for most of the evaporation process the heat transfer coefficient is proportional to the flow quality. Although other extraction strategies could be studied, it is not expected that any strategy will lead to significant reductions in required heat transfer area at least for the cases where the number of circuits is kept constant through the process. It would be interesting to study the effects of reducing the number of circuits as vapor is extracted. It would also be interesting to perform a similar study where saturated liquid is extracted from the condenser such that the quality remains high.

NOMENCLATURE

A	Cross sectional area [m^2]
D	Tube inside diameter [m]
G	Mass flux [kg/s-m^2]
g	Gravitational acceleration [m/s^2]
h	Specific enthalpy [J/kg]
h^o	Stagnation enthalpy [J/kg]
h_f	Specific enthalpy of saturated liquid [J/kg]
h_{fg}	Latent heat of vaporization [J/kg]
h_g	Specific enthalpy of saturated vapor [J/kg]
h_{tp}	Two-phase heat transfer coefficient [W/K-m^2]
h_v	Vapor phase heat transfer coefficient [W/K-m^2]
h_{wet}	Liquid phase heat transfer coefficient [W/K-m^2]
L	Total tube length [m]
l	Length of one circuit [m]
$\dot{m}$	Mass flow rate [kg/s]
$\dot{m}_f$	Mass flow rate of saturated liquid [kg/s]
$\dot{m}_g$	Mass flow rate of saturated vapor [kg/s]
P	Pressure [Pa]
p	Perimeter [m]
q''	Heat flux [W/m^2]
T	Temperature [^{o}C]
u	Speed [m/s]
v_f	Saturated liquid specific volume [m^3/kg]
v_g	Saturated vapor specific volume [m^3/kg]
x	Flow quality
z	Position [m]
Greek letters	
α	Void fraction
θ_{dry}	Dry angle [radians]
σ	Surface tension [N/m]
ρ	Density [kg/m^3]
τ	Shear stress [N/m^2]
Subscripts	
f	Saturated liquid; friction
g	Saturated vapor

REFERENCES

Kattan, N., Thome, J.R., Favrat, D., 1998, Flow Boiling in Horizontal Tubes: Part 1 – Development of a Diabatic Two-Phase Flow Pattern Map, Journal of Heat Transfer, Vol. 120, p. 140-147

Kattan, N., Thome, J.R., Favrat, D., 1998, Flow Boiling in Horizontal Tubes: Part 3 – Development of a New Heat Transfer Model Based on Flow Pattern. Journal of Heat Transfer, Vol. 120, p. 156-165

Müller-Steinhagen, H., Heck, K., 1986, A simple friction pressure drop correlation for two-phase flow in pipes, Chem. Eng. Process, Vol. 20, p. 297-308

Ould Didi, M.B., Kattan, N., Thome, J.R., 2002, Prediction of two-phase pressure gradients of refrigerants in horizontal tubes, International Journal of Refrigeration, Vol. 25, p. 935-947

Shen, B., Groll, E.A., Braun, J.E., 2005, Improvement and Validation of Unitary Air Conditioner Heat Pump Simulation Models for R-22 and HFC Alternatives at Off-Design Conditions (1173-TRP)

Steiner, D., 1993, Heat Transfer to Boiling Saturated Liquids, VDI-Wärmeatlas, Verein Deutscher Ingenoeure, ed., VDI-Gessellschaft Verfahrenstechnik und Chemieingenieurwesen

Wojtan, L., Ursenbacher, T., Thome, J.R., 2005, Investigation of flow boiling in horizontal tubes: Part I – A new diabatic two-phase flow pattern map, International Journal of Heat and Mass Transfer, Vol. 48, p. 2955-2969

Wojtan, L., Ursenbacher, T., Thome, J.R., 2005, Investigation of flow boiling in horizontal tubes: Part II – Development of a new heat transfer model for stratified-wavy, dryout and mist flow regimes, International Journal of Heat and Mass Transfer, Vol. 48, p. 2970-2985

Wojtan, L., Ursenbacher, T., Thome, J.R., 2005, Measurement of dynamic void fractions in stratified types of flow, Experimental Thermal and Fluid Science, Vol. 29, p. 383-392

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